NASA Technical Memorandum 4257

Automatic Control Study of the Icing Research Tunnel Refrigeration System

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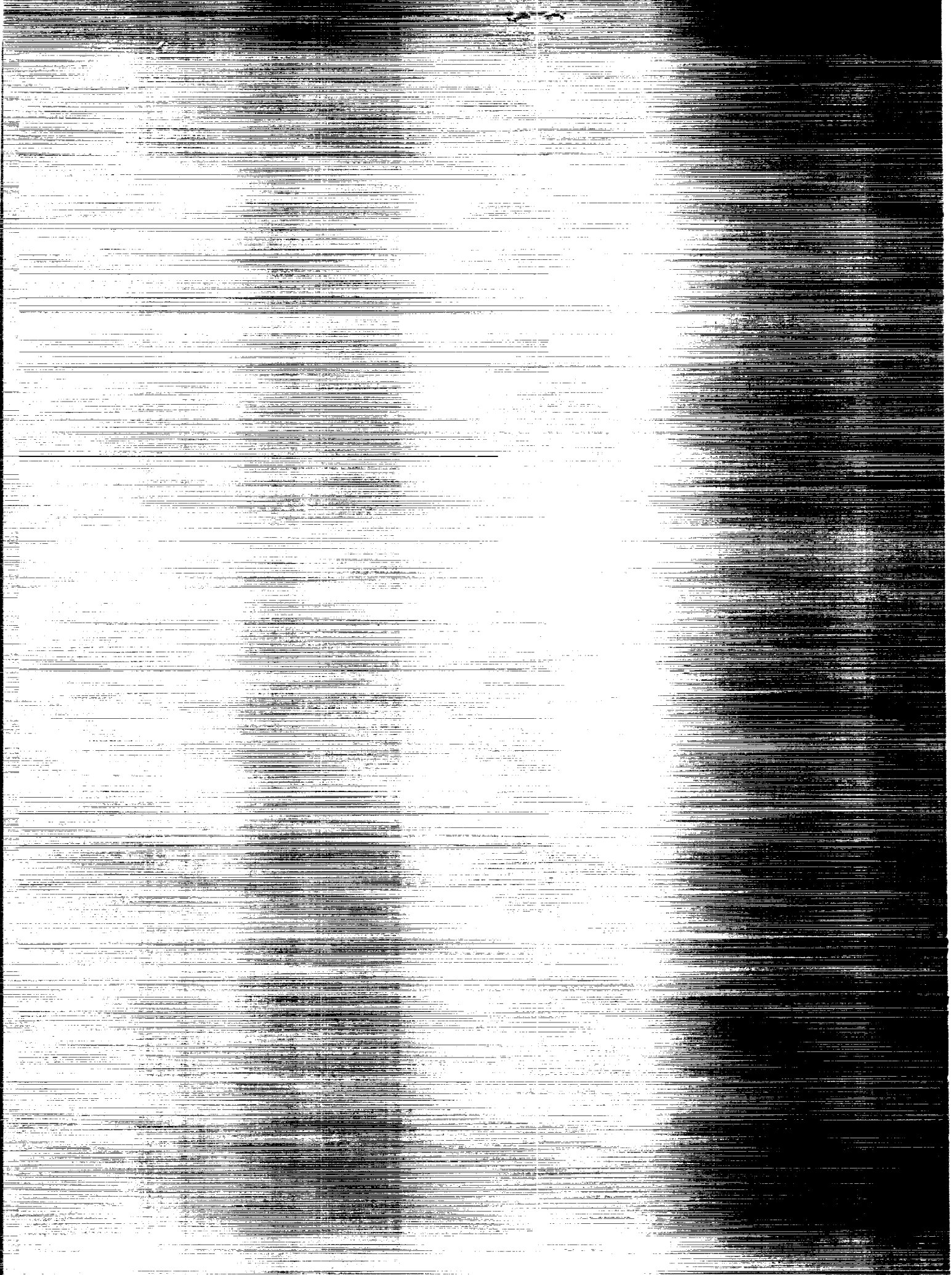JANUARY 1991

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Summary

The Icing Research Tunnel (IRT) at the NASA Lewis Research Center is a subsonic closed-return atmospheric tunnel. The tunnel includes a heat exchanger and a refrigeration plant to achieve the desired air temperature and a spray system to generate the type of icing conditions that would be encountered by aircraft. At the present time, the tunnel air temperature is controlled by manual adjustment of freon refrigerant flow control valves. An upgrade of this facility calls for these control valves to be adjusted by an automatic controller. This report discusses the digital computer simulation of the IRT refrigeration plant and the automatic controller that was used in the simulation.

Introduction

The Icing Research Tunnel (IRT) at the NASA Lewis Research Center is a closed-return, atmospheric tunnel equipped for testing low-speed aerodynamic models. A standard centrifugal compressor refrigeration cycle is used to reduce the tunnel air temperature to the desired test conditions. At present, the temperature is controlled manually by a refrigeration system operator who manipulates the freon refrigerant flow control valves. As part of an upgrading and rehabilitation of the IRT Facility, this study was performed to design an automatic control which could be used as an alternative to the manual control. Automatic control would make more efficient use of the tunnel run time. Presently it takes the system operator considerable time to change the tunnel air temperature from one operating point to another. This study demonstrates that an automatic controller could accomplish this in less time. For this investigation, the IRT refrigeration system was simulated on a digital computer. The purpose of the simulation was two-fold: (1) to test various temperature control schemes and design a suitable controller and (2) to study future facility automation in areas such as startup, shutdown, and decision-making capabilities (such as changing the number of compressors during run time).

System Description

The major components of the Icing Research Tunnel (IRT) Refrigeration System are compressors, pumps, condensers, economizers, a flash cooler, and a heat exchanger. The system removes heat from the IRT by means of a heat exchanger and maintains the tunnel air temperature at the required operating point. A standard centrifugal compressor refrigeration cycle is used, with Freon-12 as the refrigerant. The system is operated in a flooded manner where subcooled refrigerant is continually circulated through the tunnel heat exchanger. This feature improves heat transfer by keeping the refrigerant a liquid until near the end of its passage through the heat exchanger.

The refrigeration system is shown schematically in figure 1. Liquid freon is taken at low pressure and temperature from the bottom of the flash cooler and is pumped through a series of pipes (approximately 240 ft long) and headers leading to tunnel heat exchanger coils. At the design flow rate of 150 lb/sec, it takes the liquid freon approximately 50 sec to travel from the flash cooler to the heat exchanger. The heat exchanger, located inside the tunnel, consists of 80 individual heat exchanger panels arranged in a pleated design. Rectangular sheet fins attached to copper coils carry heat from the tunnel air into the copper coils, causing the freon flowing inside the coils to partially vaporize before leaving the heat exchanger and returning to the flash cooler.

In the flash cooler the liquid freon drops to the bottom of the tank to be again recirculated by the pump. The freon vapor in the flash cooler flows through damper valves into an accumulator tank. The damper valves control the tunnel temperature by regulating the flow in the compressor loop. From one to six parallel-operated damper valves can be used depending on tunnel heat load. As the freon vapor leaves the accumulator, it enters the compressor where its pressure and temperature are raised. A super-heated freon vapor then leaves the compressors and enters the tube-in-shell condensers where cooling-tower water is used to remove heat from the high-pressure, high-temperature freon vapor. The heat removed from the freon causes the refrigerant vapor to condense into a saturated liquid.
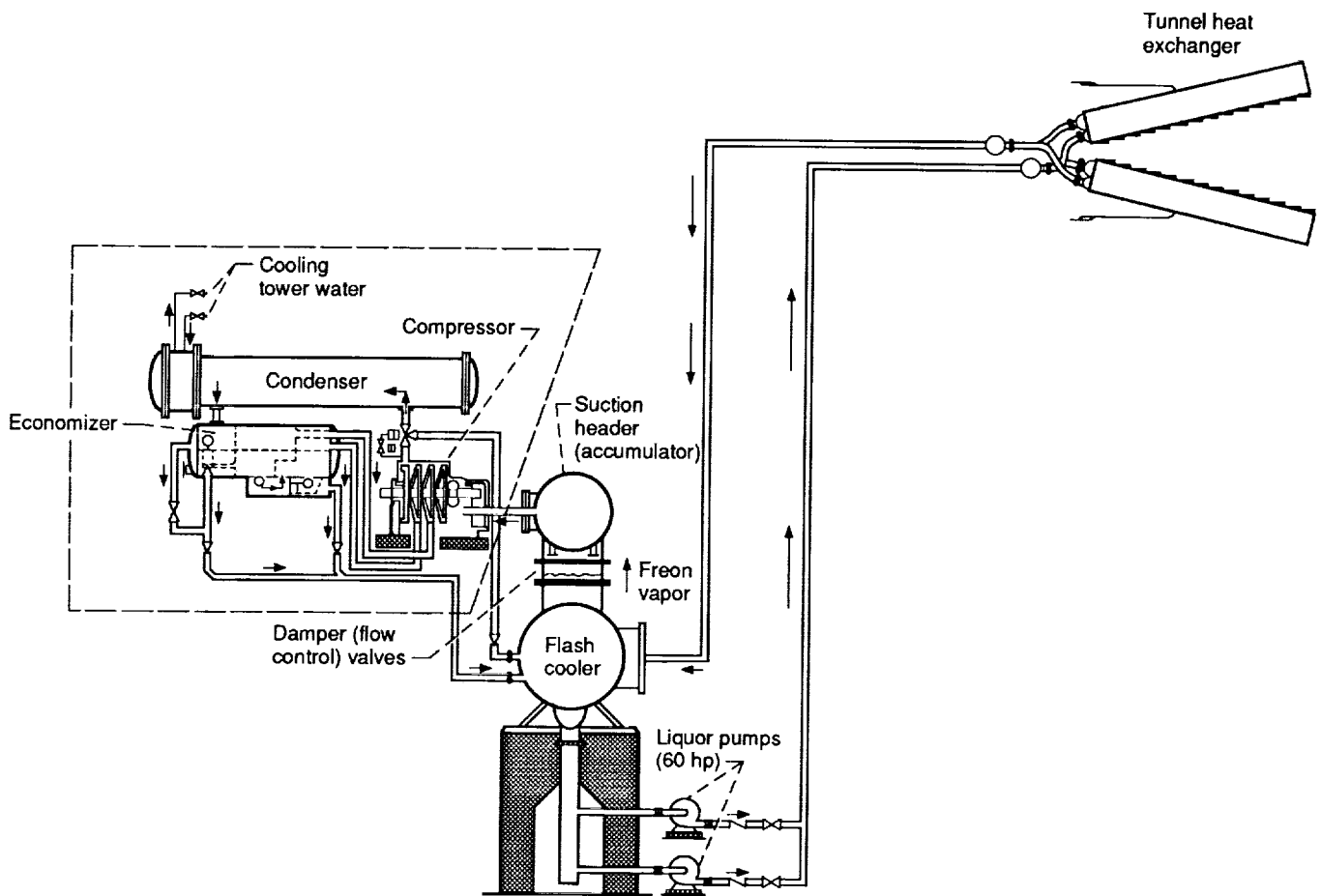
Figure 1.—Refrigeration system for Icing Research Tunnel. (Compressor-condenser-economizer unit shown inside dashed lines.)

Freon from the condenser may flow into a two-stage economizer (compressor interstage cooler) or may bypass the economizer and flow back to the flash cooler depending on the operating conditions. When the economizers are used, liquid freon enters through a set of orifices which results in a pressure drop and causes part of the fluid to flash to the vapor state. The flash vapor is routed back to the inlet of the second and third stages of the centrifugal compressors. The liquid part of the freon leaves the economizers at a low pressure and temperature and returns to the flash cooler where part of it flashes to vapor and again becomes available to be drawn through the damper valves for a restart of the vapor cycle. The liquid part of the refrigerant in the flash cooler is available to be pumped to the tunnel heat exchanger for tunnel heat removal.

The four-stage centrifugal compressor, a two-pass tube-in-shell water-cooled condenser, and a two-stage economizer (used to improve the efficiency of the process) are all part of a self-contained unit. One compressor-condenser-economizer unit is shown schematically within the dashed lines of figure 1. A 1500-hp induction motor drives each unit. At present the refrigeration system contains 13 such units which can be used during a run.

Computer Model

Model Description

A computer model was made of the IRT Refrigeration System and was used to formulate and test the IRT temperature control design presented here. This model can be used for further automatic control studies in areas such as startup and shutdown.

The computer model uses algebraic and time differential equations to describe steady-state as well as transient conditions. A lumped-parameter approximation of the system components is made using control volumes. Transient equations of conservation of mass, conservation of energy, and equations of state are used to solve for the temperature and pressure for these components. Appropriate flow equations for the vapor flow through the damper valves (control valves), condensation flow in the condenser, and evaporative flow in the flash cooler and economizer are used. Line pressure drops, and transients that have no effect on the control design were neglected. A schematic of the computer model is shown in figure 2. A detailed description of the refrigeration system computer model with equations is presented in appendix A.
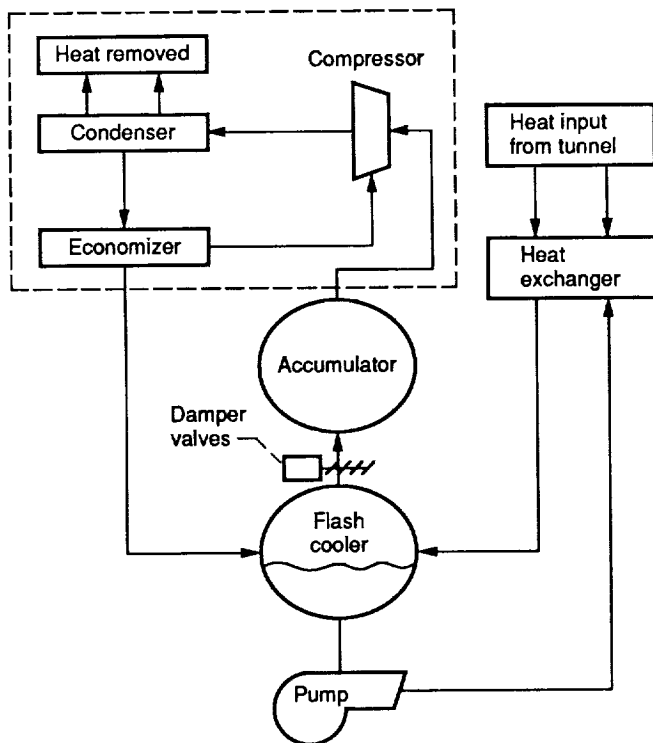
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Figure 2.—Simulation diagram for Icing Research Tunnel. (Compressor-condenser-economizer unit shown inside dashed lines.)

The computer simulation was programmed in the CSMP3 simulation language. This simulation language is well suited for modeling systems that can be described by ordinary time-dependent nonlinear differential equations.

Model Validation

The refrigeration system was instrumented for pressure and temperature at various points of interest primarily to obtain a valid compressor characteristic curve at the 6650-rpm design operating point. These temperature and pressure data were also compared with the predicted steady-state results of the computer model.

Table I compares data from a tunnel run with those from a similar computer run. The two runs closely agree for all

TABLE I.—TUNNEL DATA COMPARED WITH COMPUTER RESULTS

Variable	Tunnel run data	Computer results
Temperatures, °R		
Tunnel	460	460.0
Flash cooler	448	454.6
Compressor inlet	453	454.5
Compressor exit	679	676.7
Water exit	529	530.8
Pressures, psia		
Flash cooler	19.3	21.2
Compressor inlet	8.3	8.8
Compressor exit	99.0	98.1
Compressor motor current, A	250	175.2
Compressor loop flow, lb/sec	[a]24	23.6

[a]Calculated from data.

variables except the compressor motor current. A much higher motor current was needed in the actual tunnel run than was predicted by the computer model. This indicates that the compressor-motor unit was probably operating at a much lower efficiency (40 percent) than was used in the simulation (70 percent). This was not surprising since the system was operating at off-design conditions.

Control Development

System Characteristics

The IRT Refrigeration System is not well suited for automatic control mainly because of the numerous operating constraints of the system:

(1) Operating constraint caused by compressor surge
(2) Upper limit of compressor inlet (accumulator) pressure of 12 psia (because of seal limitations)
(3) Upper limit of condenser pressure of 125 psia (because of structural limitations)
(4) Upper limit of compressor motor current of 300 A

Compressor surge and compressor inlet pressure are the most frequently encountered constraints. They can be exceeded whenever the tunnel operating temperature is changed too rapidly. The compressor can surge when the tunnel temperature is increased, and the compressor inlet pressure can exceed 12 psia when the tunnel temperature is decreased. This study focused primarily on the compressor inlet pressure constraint because the compressor surge was not well defined and will not be as much of a problem.

Controller Description

The controller was designed to bring the tunnel air to the desired operating temperature as fast as possible without exceeding any system constraints. This was best accomplished by changing the temperature at a constant rate. Moreover, to minimize time delays, the controller was designed to bring the temperature to its new steady-state value with little over-shoot and minimum settling time.

Standard control analysis techniques were not used in this study because the system is nonlinear. Instead, various temperature control schemes were tried on the computer. In the scheme that was selected, the outer control loop controlled the tunnel temperature, an intermediate loop regulated the flash cooler temperature, and an inner loop controlled the valve position.

The flash cooler temperature is used as the intermediate loop because it closely approximates the tunnel temperature in steady state and responds immediately to a change in control valve position. The tunnel temperature responds more slowly because the large thermal capacity of the tunnel heat exchanger introduces considerable temperature lag between the flash cooler and tunnel temperatures.

In addition to the thermal lag, there is a temperature transport lag caused by the fluid flowing into the heat exchanger.
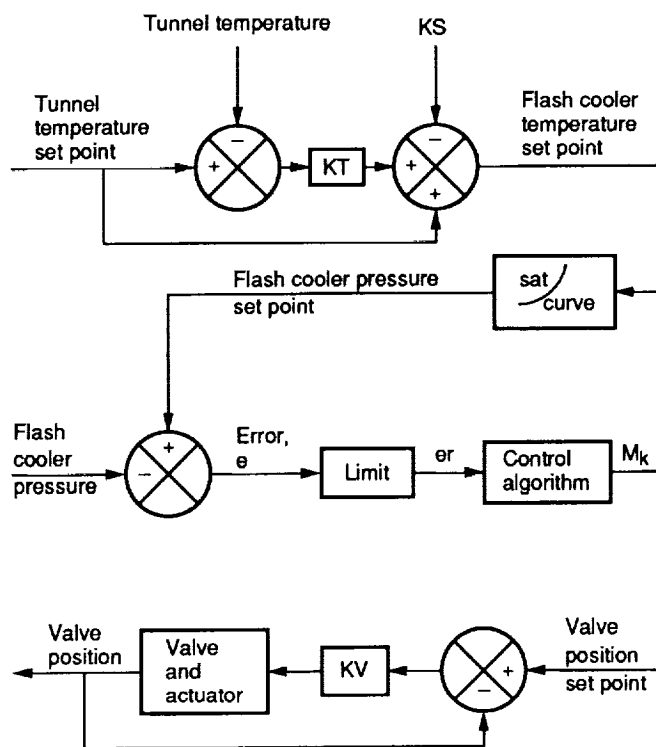
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Figure 3.—Tunnel temperature controller.

It takes 50 sec for the liquid freon to travel from the flash cooler to the heat exchanger. This long dead time, if it were a predominant effect, would be a serious hindrance in the design of an effective automatic temperature controller. However this is not the case here because any immediate freon temperature change is due to a change in control valve position. The valve position, which affects the flash cooler pressure and hence flash cooler temperature, also affects the heat exchanger temperature because both pressures are essentially the same. Therefore the temperature of the freon in the heat exchanger changes at the same time as the flash cooler temperature and is not dependent on the fluid flowing in from the flash cooler. The temperature of this fluid should have only a minor effect on the temperature transients in the heat exchanger. The computer model therefore neglects this transport delay.

The setpoint for the flash cooler temperature control is generated by the outer control loop. At steady state, the flash cooler temperature generally is 5° lower than the tunnel temperature. Therefore to account for this difference, the flash cooler temperature setpoint was made 5° lower than the tunnel temperature setpoint. Because the difference is not quite constant and may change somewhat for different operating conditions, a proportional temperature control was added. The proportional controller not only corrects for the steady-state temperature difference, but also speeds up the control response.

Because the flash cooler fluid is saturated, its vapor pressure is strictly a function of temperature. As a result, the flash cooler temperature control can be converted to a pressure control. This conversion should improve the controller response

because the flash cooler pressure response is faster than the temperature response. Also, pressure control may be more desirable than temperature control, from a measurement perspective.

Valve position was added as an inner control loop so that the system could be switched to manual control. Since the dynamics of the inner loop are much faster than those of the other components in the temperature controller, the inner loop has essentially no effect on the overall response.

A block diagram of the Tunnel Temperature Controller is presented in figure 3. The KT block represents a gain constant for the outer temperature control loop. The KS block represents a correction for the steady-state temperature differences between the flash cooler temperature and the tunnel temperature. The "sat curve" block represents a lookup table which converts freon temperature to pressure. The "LIMIT" block limits the flash cooler pressure error signal before it enters the Control Algorithm block. The block output M_k represents the control valve position setpoint. The KV block is a gain constant for the valve position control loop. The equations for the control algorithm of the pressure control loop follow.

Control Algorithm

The control algorithm is depicted in the block diagram shown in figure 3 as follows:

$$M_{k-2} = M_{k-1}$$
$$M_{k-1} = M_k$$
$$er_{k-2} = er_{k-1}$$
$$er_{k-1} = er_k$$

Test for setpoint step change; if step, reset

$$er_{k-1} = er_{k-2} = 0 \quad \text{and} \quad M_{k-1} = M_k + FM$$

Then continue

$$FM = G(-0.799\ er_{k-1} + 1.998\ er_{k-2})$$
$$M_k = 1.3M_{k-1} - 0.3M_{k-2} + Ger_k + FM$$

where G is a controller gain which is variable and a function of tunnel heat load, the control valve position, and the number of compressors in operation; er is the control algorithm input; M is the control algorithm output; the subscript $k-n$ represents the $(k-n)^{th}$ sampling instant. The algorithm of the pressure control loop was not tuned for tunnel pressure response, but rather for satisfactory tunnel temperature response.

Control Run Results

The controller was tested at various operating conditions and for various setpoint changes. When the controller is implemented in the field, the tunnel temperature changes will probably be made with step changes in the temperature

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TABLE II.—NOMINAL OPERATING CONDITIONS
[Step setpoint change.]

Initial temperature, °R	490
Tunnel heat load, Btu/sec	1200
Compressor used	1
Control valves used	3
Evaporation constant	10.0
Control constants	
Error limit	8.0
Sample rate, samples/sec	2
Valve loop gain, KV	10.0
Outer loop gain, KT	15.0

setpoint. Consequently, all but one of the computer runs that are presented are for step setpoint changes of various sizes and at various operating temperatures.

The evaporation rate in the flash cooler and the heat transfer coefficient in the tunnel heat exchanger were two parameters whose values were considered to be very uncertain. Since these parameters were deemed important to the system response, control runs were made where these parameters were varied. Other important system parameters were also investigated: the tunnel heat load, the number of compressor units in operation, the number of control valve units being used, the effect of controller gain (change in the limit of the pressure error signal), and controller sampling rate. Because of the uncertainty of the compressor characteristic curve, a control run was made where the slope of the compressor curve was increased by 100 percent to determine its effect on the closed-loop response.

A typical tunnel operating condition and controller setting were selected and termed as the nominal case. Nominal operating parameters are presented in table II. The time response for runs with different operating conditions are compared with this nominal case whenever applicable.

Run 1—Nominal Case

The first run presented is for the nominal case with the tunnel operating at a temperature setpoint of 490 °R. At 80 sec after
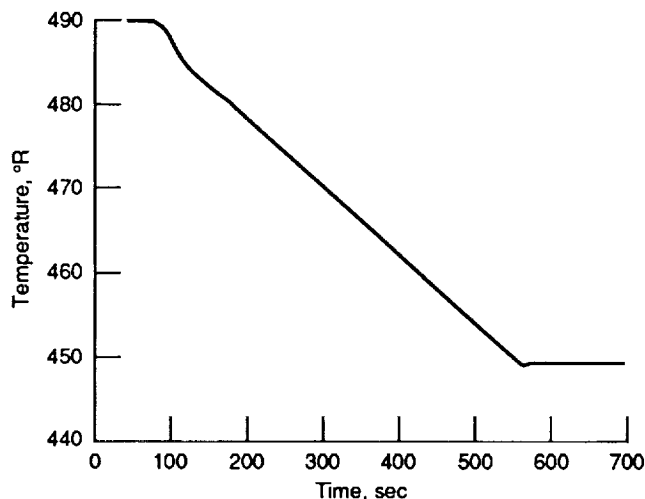

Figure 4.—Tunnel temperature response for setpoint decrease from 490 to 450 °R.

the start of the run, the tunnel temperature setpoint was stepped from 490 to 450 °R. The controller performed according to design expectations in that the tunnel temperature response (fig. 4) was similar to a ramp and (very importantly) the response showed no overshoot. It took 485 sec (from 80th to 565th sec) for the tunnel temperature to reach its new setpoint of 450 °R. The control valve movement for the run is presented in figure 5. The valve started from an initial position of approximately 2° and moved to a final position of approximately 4.7°. The compressor inlet (accumulator) pressure is a critical parameter in these runs because its upper limit of
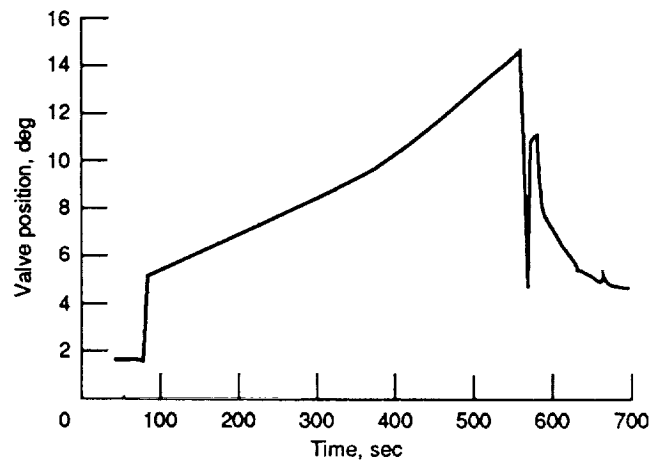

Figure 5.—Control valve movement for tunnel temperature setpoint decrease from 490 to 450 °R.
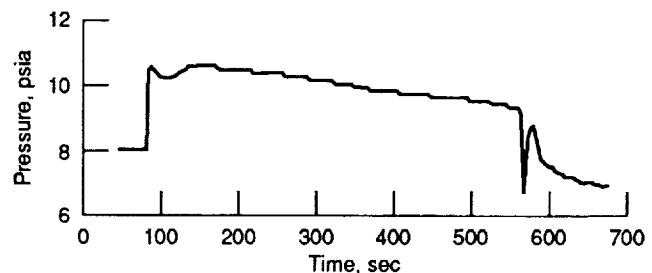

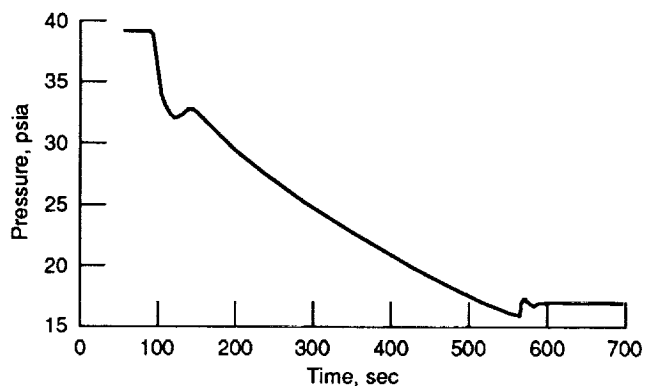Figure 6.—Accumulator pressure for tunnel temperature setpoint decrease from 490 to 450 °R.

Figure 7.—Flash cooler pressure change for tunnel temperature setpoint decrease from 490 to 450 °R.

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12 psia can often be exceeded during the transients. The accumulator pressure (fig. 6) was initially 8.1 psia. At the start of the transients it reached a peak of 10.6 psia and then settled to a new steady-state value of 7.1 psia. The 12-psia limit was not exceeded during this run. The flash cooler pressure is provided in figure 7 as a matter of interest.

Run 2—Compressor Change

Run 2 was identical to run 1 except that two compressors were used instead of one. This change increased the tunnel temperature response time by about 45 sec (fig. 8). The temperature reached steady state at about 520 sec with a slight overshoot. (In run 1, there was no overshoot.) As was expected, the system responded faster when more compressors were added to the system.

Run 3—Control Valve Change

Run 3 was also identical to run 1 except that four control valve units were used instead of three. The tunnel temperature (fig. 9) reached steady state at approximately 450 sec with no overshoot. The response is at least 100 sec faster than the nominal case (fig. 4) of run 1.

Run 4—Change in Operating Point

Because the system is nonlinear, run 4 was made to show controller performance at other operating points. Run 4 started
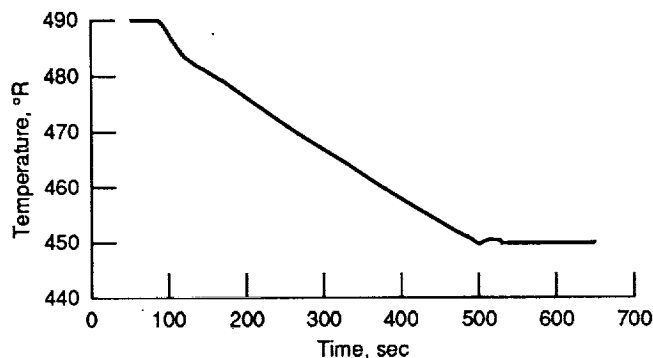

Figure 8.—Tunnel temperature response for tunnel temperature setpoint decrease from 490 to 450 °R with two compressors.
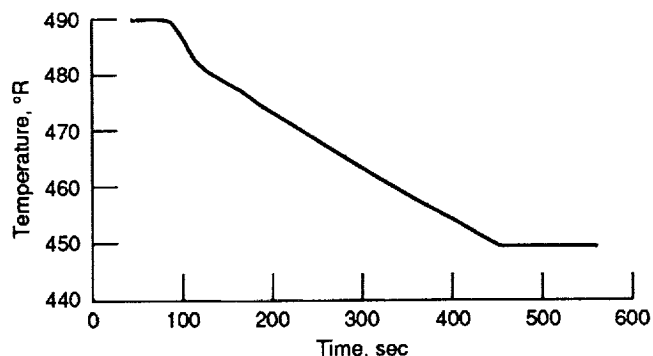

Figure 9.—Tunnel temperature response for tunnel temperature setpoint decrease from 490 to 450 °R with four damper valves.
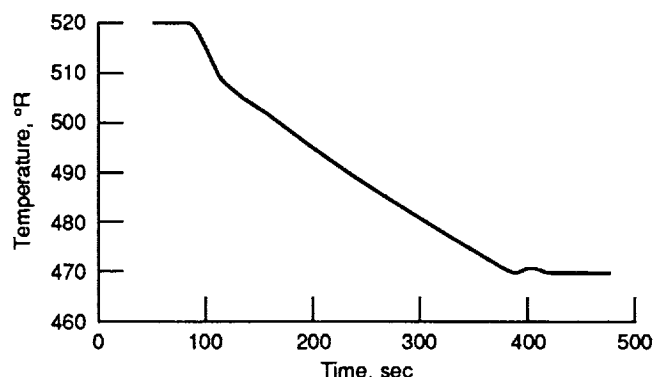

Figure 10.— Tunnel temperature response for tunnel temperature setpoint decrease from 520 to 470 °R.
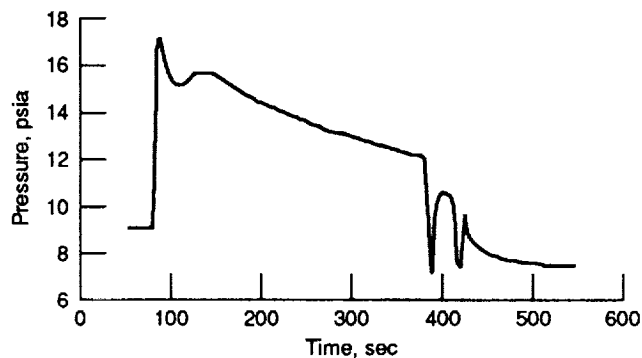
Figure 11.—Accumulator pressure change for tunnel temperature setpoint decrease from 520 to 470 °R.

Figure 12.—Accumulator pressure change for tunnel temperature setpoint decrease from 520 to 470 °R with controller gain reduced by 50 percent.

from an initial operating point of 520 °R. At 80 sec the setpoint was stepped from 520 to 470 °R. The tunnel temperature response (fig. 10) showed a slight overshoot. Even though the setpoint change was 10° larger for this run than for run 1, the response is more than 140 sec faster because the required change in cooling power is smaller at the higher temperatures. One other observation is noteworthy. The higher tunnel operating temperature generally raised the pressure and temperature of the entire refrigeration system, increasing the likelihood of the accumulator pressure limit being exceeded during transients. Figure 11 shows the accumulator pressure. It exceeds its 12 psia limit and briefly rises above 17 psia.

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Figure 13.—Tunnel temperature response for tunnel temperature setpoint decrease from 520 to 470 °R with controller gain reduced by 50 percent.

Figure 14.—Tunnel temperature response for tunnel temperature setpoint ramped from 520 to 470 °R in 100 sec.

Figure 15.—Tunnel temperature response for tunnel temperature setpoint decrease from 460 to 440 °R.

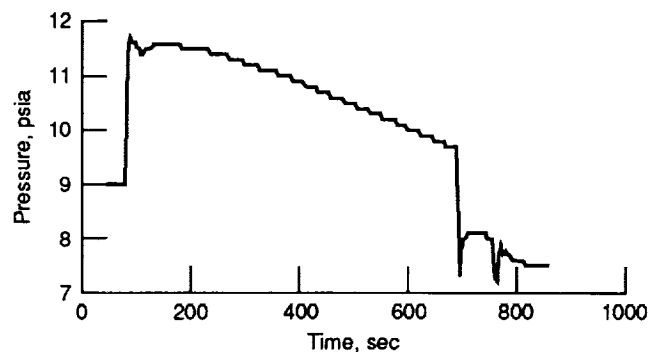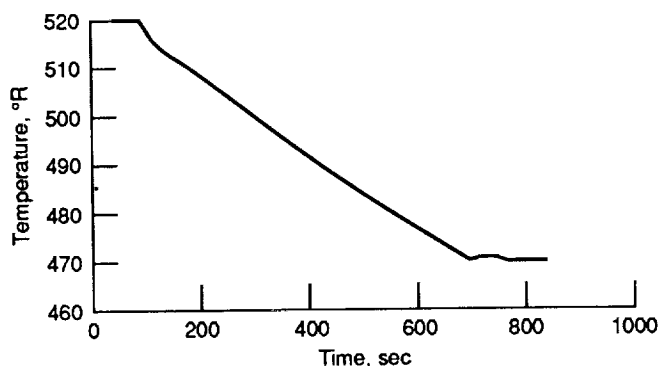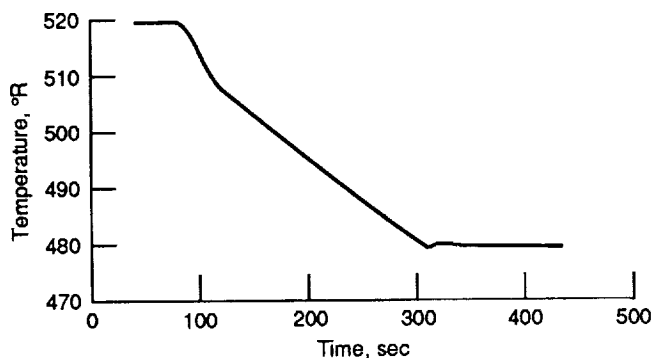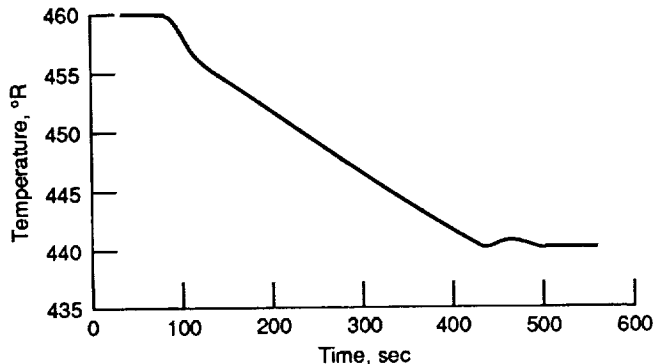## Run 5—Controller Gain Change

A lower controller gain is needed to keep the accumulator pressure in run 4 below its limit. Consequently, run 4 was repeated with the controller gain (limit on the pressure error signal) lowered by 50 percent. The reduced gain kept the accumulator pressure transient below the 12-psia limit (fig. 12), but it also produced a slower temperature response (fig. 13; about 340 sec slower than in run 4). The gain will probably not have to be lowered, for this case, because a refurbished set of compressors should be installed by the time

this facility is automated. The new accumulator pressure limit will then be approximately 20 psia.

Run 6—Setpoint Ramp

In the previous runs all temperature setpoint changes were step changes. To test the controller for a ramp setpoint change, run 6 was made with the tunnel temperature setpoint ramped from 520 to 480 °R in 100 sec. The tunnel temperature response (fig. 14) is similar to a step setpoint change.

Run 7—Small Setpoint Change (Actual Tunnel Run Comparison)

A small setpoint change was made from 460 to 440 °R at 80 sec. It takes 380 sec or 6.3 min for the temperature to reach steady state at its new operating point (fig. 15). This run was compared with data from an actual run where the operator controlled the tunnel temperature manually. The system operator took 20 min to manually change the tunnel temperature from 460 to 440 °R. This is more than three times longer than by automatic control.

Run 8—Heat Load Ramp

The tunnel fan speed is often changed during tunnel operation to accommodate wind tunnel test requirements. Because the fan is the major tunnel heat source, changing the fan speed also changes the tunnel heat load. The effect of this heat load change on the temperature control was studied by ramping the heat load from 800 Btu/sec (at 100 sec) to 1500 Btu/sec (at 200 sec). The setpoint was held at 460 °R during the ramp. The automatic controller was able to maintain the tunnel temperature near its setpoint with essentially no temperature change (fig. 16). Figure 17 shows the valve position change
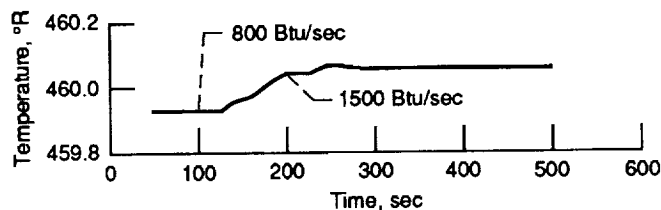

Figure 16.—Tunnel temperature variation during a tunnel heat load increase of 700 Btu/sec.
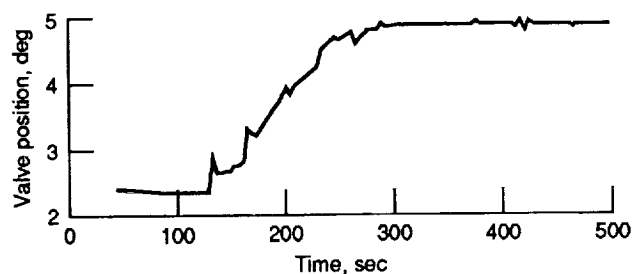

Figure 17.—Control valve position during a tunnel heat load increase of 700 Btu/sec.

made by the controller in order to maintain the temperature at its setpoint during the heat load ramp. The valve moved from an initial position of approximately 2.4° to a final position of 4.9°.

Run 9—Setpoint Increase

The general operating procedure for all tunnel runs is to start at ambient conditions and then decrease the tunnel temperature to the desired operating point. Therefore, all of the computer runs presented thus far were for decreases in the temperature setpoint. Occasionally, however, the tunnel temperature needs to be increased to a new operating point. Figure 18 shows such a computer run where the temperature setpoint is increased from 450 to 490 °R at 700 sec into the run (run 9). The quality of the response is similar to the step decreases shown in the previous runs. For this run it takes 690 sec for the temperature to go from 450 to 490 °R. There is no overshoot. During this excursion the accumulator pressure transient moves away from the 12-psia limit. This is typical for all runs where the tunnel temperature is increased. Because the accumulator pressure is not a problem for these runs, it is not examined when the tunnel temperature is increased. Compressor surge, however, can be a problem for large, rapid tunnel temperature increases. The problem was investigated; but because the compressor surge line for the present compressors is not well defined, the run results from this investigation are not presented here. If compressor surge becomes a problem for the automated system, the controller gain will have to be reduced for certain runs involving tunnel temperature increases.

The response to a setpoint increase in the tunnel temperature is often limited by the present configuration of the refrigeration system. For this system, the control valve usually needs to be operated near its closed position—a practice that restricts any valve movement in the closed direction and, consequently, limits the response for tunnel temperature increases. The valve movement for run 9 is shown in figure 19. Prior to the setpoint
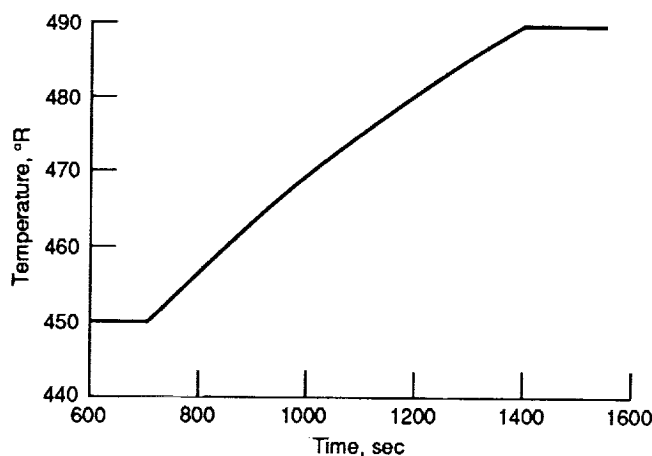

Figure 18.—Tunnel temperature response for tunnel temperature setpoint increase from 450 to 490 °R.
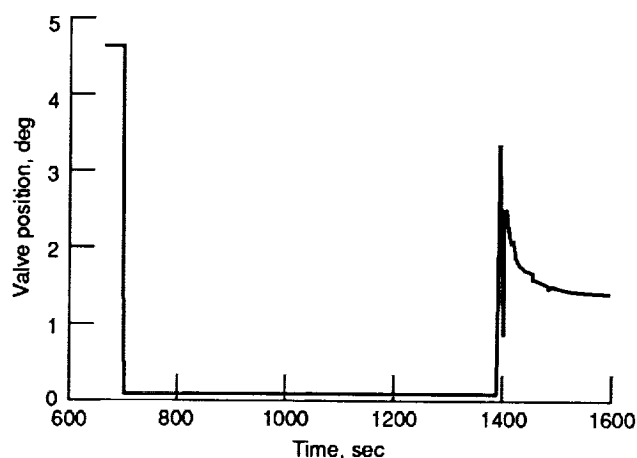
Figure 19.—Control valve position for tunnel temperature setpoint increase from 450 to 490 °R.

increase, the valve position was 4.6°. As soon as the setpoint increased, the valve nearly closed and remained so until the tunnel temperature had almost reached the new setpoint value. At that time, the valve moved to a final steady-state position of 1.4°.

Runs 10 to 13—Additional Runs With Setpoint Increases

Other runs tested different temperature setpoint increases similar to that in run 9. The tunnel temperature response for each of these runs was very satisfactory with essentially no overshoot or oscillation as is shown in figure 20. The operating conditions for these runs were similar to the previously presented runs.

Small Setpoint Disturbances

The next set of runs were made to further demonstrate the robustness of the controller and to show the effect of various parameters on closed-loop performance. For these runs the setpoint was stepped from 460 to 440 °R. Run 7 will be considered as the reference or nominal case, and all parametric variations of the controller and the system will be compared to figure 15 of run 7.

Controller Variation Runs

Figure 21 shows the effect of two important controller settings on the temperature response. First, the controller sampling rate was reduced from 2 samples per sec to 1 sample per sec (the limit on the error signal was doubled to offset the gain change caused by the sampling rate). The sampling rate change showed virtually no effect on the system. Next, the limiter output of the error signal was changed from its nominal value of 8 to a value of 12 (see fig. 3). The temperature shows no overshoot and reaches its setpoint value about 80 sec sooner and its steady-state value almost 150 sec sooner than for the nominal gain setting.

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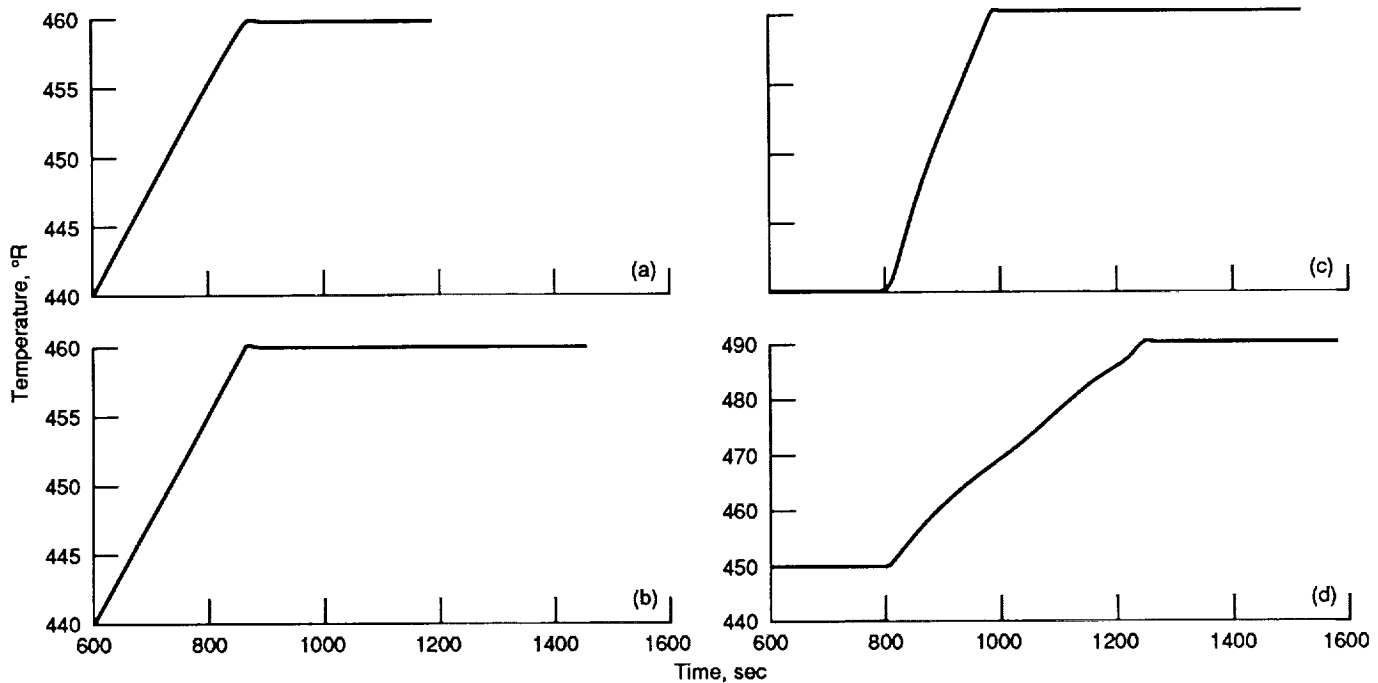
(a) Setpoint increase from 440 to 480 °R at nominal operating conditions.
(b) Setpoint increase from 440 to 460 °R with controller gain at 150 percent.

(c) Setpoint increase from 440 to 460 °R with three compressors and a tunnel heat load of 2000 Btu/sec.
(d) Setpoint increase from 450 to 490 °R with tunnel heat load at 2000 Btu/sec.

Figure 20.—Tunnel temperature response for various tunnel temperature setpoint increases.

System Variation Runs

Two system parameters which were thought to be important, and yet whose values are very uncertain, are the evaporation rate in the flash cooler and the heat transfer coefficient in the tunnel heat exchanger. A large part of the heat transfer coefficient uncertainty comes from the approximate calculation of the fluid quality in the heat exchanger. Figure 22 shows the temperature response for a nominal value of the evaporation rate constant, KVE, and for a 100 percent increase of KVE. The increase had little effect on the temperature response. The temperature response increased slightly when

the heat transfer coefficient was changed to twice its nominal value, but the results are not presented here.

Heat Load Variation Runs

The tunnel heat load is mostly a function of tunnel fan speed (rpm) and, to some extent, the outside ambient temperature. Figure 23 shows the controller performance for different tunnel heat loads—1200 (nominal), 2000, and 800 Btu/sec. As would be expected, the lowest heat load shows the best response.
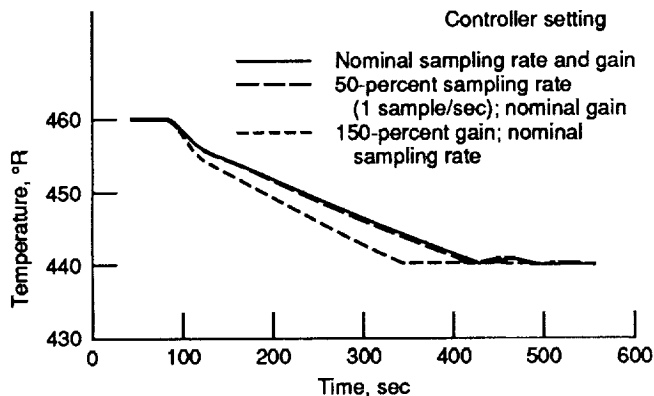

Figure 21.—Tunnel temperature response for various controller settings (sampling rate and gain).
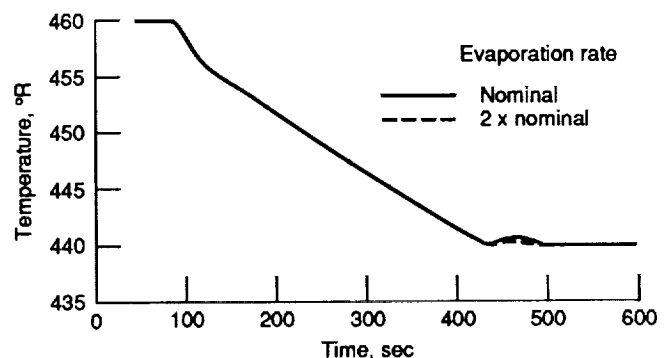

Figure 22.—Tunnel temperature response to variations in evaporation rate.
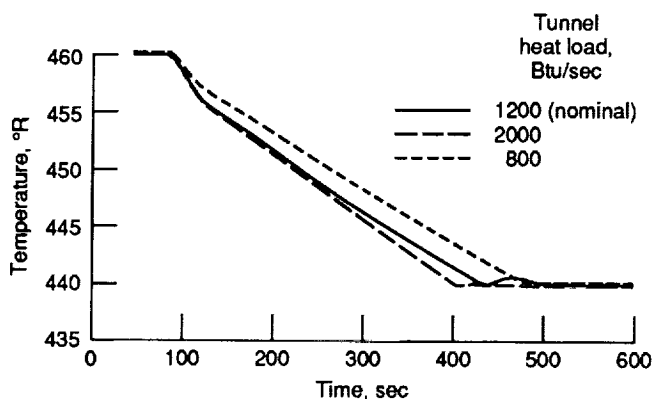
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Figure 23.—Tunnel temperature response for various tunnel heat loads.

Compressor Variation Runs

Because of the uncertainty in the compressor characteristic curve, and the likelihood that a refurbished compressor will be installed in the system, a control run was made where the slope of the current compressor characteristic curve was increased by 100 percent (values for compressor head were doubled). The temperature response with the new compressor slope is shown in figure 24. The response shows no overshoot. The temperature reaches its steady state approximately 30 sec sooner than for the run with the nominal compressor characteristic curve.

Noise Injection

The controller was also tested with Gaussian noise added to the control signals of the flash cooler and to the tunnel temperature. The added noise was considerably higher than can ever be expected in the actual instrumentation. Since its effect on the controller was essentially negligible, the results are not presented here.
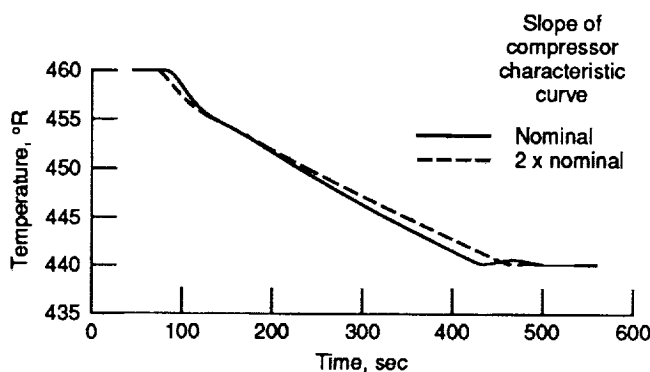

Figure 24.—Tunnel temperature response to a change in compressor characteristic curve.

Concluding Remarks

Although the Icing Research Tunnel (IRT) was designed to operate at temperatures below 440 °R, most tunnel runs are currently made above 440 °R. Consequently, operating constraints are frequently encountered. These constraints are

(1) Operating constraint caused by compressor surge
(2) Upper limit of compressor inlet (accumulator) pressure of 12 psia
(3) Upper limit of condenser pressure of 125 psia
(4) Upper limit of compressor motor current of 300 A

The most frequently encountered operating constraint appears to be the upper limit of the compressor inlet pressure. To avoid this constraint, an additional compressor unit is brought on line. This allows the control valves to operate at a more closed position, thereby causing a larger pressure drop and, consequently, a lower compressor inlet pressure. Since this practice brings more compressors on line than would otherwise be needed, it reduces the efficiency of the operation, and frequently steam must be injected into the tunnel to maintain the desired tunnel temperature. The compressor inlet pressure also poses a problem for the automatic temperature control. The controller must be designed so that this pressure constraint is not exceeded during control transients.

The primary incentive for adding an automatic temperature control is to make more efficient use of the tunnel run time. Consequently, the automatic temperature control was designed to reach the desired operating point as quickly as possible without creating excessive transients that would exceed system constraints. The objective was to ramp the tunnel temperature to its desired value with a constant rate and little overshoot. For the runs that were made in this study, the control design met this objective. Study results showed that the automatic temperature controller can change the tunnel temperature in less than half the time needed by manual control. We compared a computer run with an actual, manually controlled tunnel run. For the computer run, the automatic control lowered the tunnel temperature from 460 to 440 °R in only 6.3 min, whereas for the actual run with manual control, 20 min were needed to make the same change. The automatic controller, however, frequently caused a transient, excessive compressor inlet pressure when the controller gain was set too high.

System constraints could be avoided if automatic control were used for selected runs only—when there is no danger of exceeding system operating limits. The alternative would be to use automatic control at all times but with a low controller gain. This would avoid the large transients which exceed the operating limits. However, with a low gain, the system would respond slowly, and the tunnel run time would not be significantly reduced compared with manual operation. The only benefit of using an automatic control with the low gain option, would be to free the system operator from the task of manually controlling the tunnel temperature.

In the interest of saving tunnel run time and making the compressor operation more efficient, automatic temperature control should be added to the facility, and the facility should be modified to allow a higher compressor inlet pressure. Currently the compressors are being overhauled to make them more suitable for higher tunnel temperature runs. The overhauled compressors will permit operation at a suction header (accumulator) pressure of up to 20 psia. No major change in the control should be required for operation with the overhauled compressors.

The automatic control study presented here does not address an automatic control design for startup and shutdown or any decision-making capabilities whereby the computer would take the necessary action to avoid the various operating constraints.

These decisions and tasks will still have to be performed by the system operator. It is recommended, however, that, after successfully implementing this initial phase of the automatic control, automation of these remaining tasks should also be considered. The present computer model was made with sufficient detail to allow these additional control studies to be performed with minimal changes.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, August 13,1990

Appendix A
Computer Simulation Equations of the IRT Refrigeration System

Refrigeration Cycle

The refrigeration cycle for the compressor loop is depicted in figure 25. Saturated gas in the flash cooler (evaporator) moves through the control valves to the accumulator, or compressor inlet. The gas is then compressed to the superheated region at the compressor outlet. From the compressor outlet the gas flows into the condenser region, where heat is removed and the gas condenses to a saturated liquid. The liquid then flows into the economizer region (fig. 25 depicts a two-stage economizer), where it loses more heat at a reduced pressure and temperature. From there it flows back to the flash cooler. A diagram of the computer model is shown in figure 26.

Basic Computer Model Equations

The following basic fluid flow and energy equations are used repeatedly throughout the simulation. The equations are based on a control volume concept of finite difference equations, where i and o represent flow into and out of the control volume, respectively.

Continuity equation

$$\frac{dM}{dt} = W_i - W_o \tag{1}$$

Energy equation

$$\frac{dE}{dt} = W_i H_i - W_o H_o + \dot{Q} \tag{2}$$

where the total enthalpy E is defined as the mass M times the enthalpy H, or

$$E = MH$$
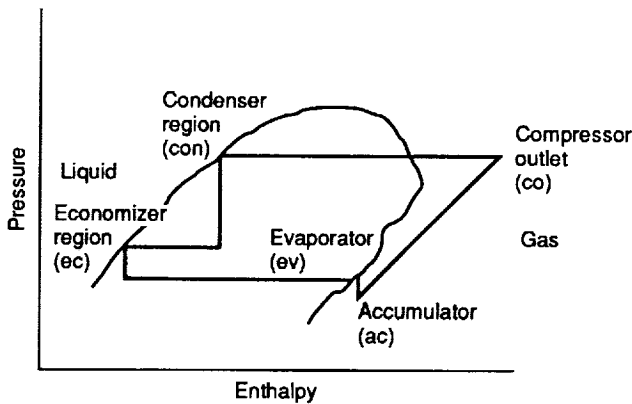
Figure 25—Refrigeration cycle.
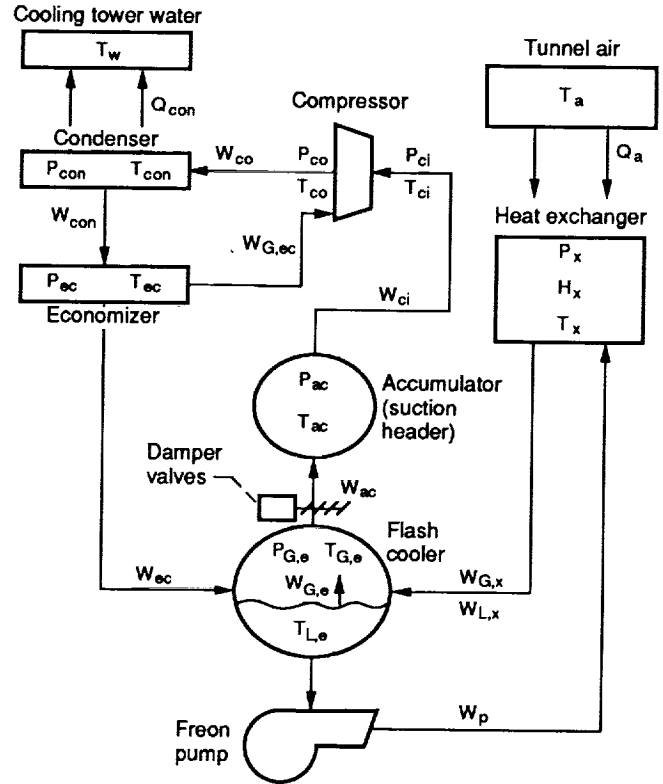
Figure 26—Simulation diagram.

and $\dot{Q}$ represents external heat conducted into or out of the control volume. For convective heat transfer,

$$\dot{Q} = hS(T_{fluid} - T_{wall}) \tag{3}$$

where

h heat transfer coefficient
S heat transfer area

Since $dH = c_p\, dT$, and if the term dM/dt can be assumed to be negligible, an alternate form of equation (2) is

$$\frac{dT}{dt} = \frac{W_i T_i - W_o T_o + \dfrac{\dot{Q}}{c_p}}{M} \tag{2a}$$

Perfect gas law

$$P = \rho RT \tag{4}$$

Orifice equation

$$W = 0.6A\sqrt{\frac{2g_c P_i (P_i - P_o)}{ZRT_i}} \tag{5}$$

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Evaporation equation[1]

$$W = KVE(P_{sat} - P_G) \qquad (6)$$

where

$$KVE = KrA\sqrt{\frac{g_c}{2\pi RT_L}}$$

The preceding equations are applied in the mathematical description of the system components presented next.

Flash Cooler Equations (Evaporator)

Energy equation.—From the energy equation (2), we have the following differential equation for the flash cooler (evaporator) liquid, where N_c accounts for the number of compressor units in operation:

$$\frac{d(E_{L,e})}{dt} = N_c W_{L,ec} H_{L,ec} + W_{L,x} H_{L,x}$$
$$- W_p H_{L,e} - W_{G,e} H_{V,e}$$

Since $W_p = W_{L,x} + W_{G,x}$ and $H_{L,x} = H_{L,e}$ because the pressure of the freon in the heat exchanger and the flash cooler are essentially the same, we can simplify the above equation as follows:

$$\frac{d(E_{L,e})}{dt} = N_c W_{L,ec} H_{L,ec} - W_{G,e} H_{V,e} - W_{G,x} H_{L,e}$$

$$H_{L,e} = \frac{E_{L,e}}{M_{L,e}}$$

Likewise, the energy equation for the flash cooler gas is

$$\frac{d(E_{G,e})}{dt} = W_{G,x} H_{G,x} - W_{ac} H_{G,e} + W_{G,e} H_{V,e}$$

$$H_{G,e} = \frac{E_{G,e}}{M_{G,e}}$$

From the saturation curve on the pressure-enthalpy (*PH*) diagram for freon (fig. 25), the freon liquid temperature is given by

$$T_{L,e} = f(H_{L,e})$$

---

[1]See Derivation of Evaporation Equation, p. 19.

Likewise, the freon gas temperature can be obtained from the saturation curve of the *PH* diagram

$$T_{G,e} = f(H_{G,e})$$

Also, the enthalpy of the evaporating freon can be obtained from the freon saturation curve:

$$H_{V,e} = f(T_{L,e})$$

And again, since the pressure of the freon in the heat exchanger and the flash cooler are essentially the same,

$$H_{G,x} = H_{V,e}$$

Continuity equation.—From the continuity equation (1), we have for the liquid

$$\frac{d(M_{L,e})}{dt} = W_{L,ec} + W_{L,x} - W_{G,c} - W_p$$

and for the gas we have

$$\frac{d(M_{G,e})}{dt} = W_{G,x} + W_{G,e} - W_{ac}$$

Equation of state.—From the definition of density we have for the density of the gas $\rho_{G,e} = M_{G,e}/V_{G,e}$, and from the density of the liquid, $V_{L,e} = M_{L,e}/\rho_{L,e}$. The flash cooler consists of both gaseous and liquid freon. Therefore the total flash cooler volume $V_e = V_{G,e} + V_{L,e}$. Using these three relations, we can now solve for the density of the flash cooler gas as follows:

$$\rho_{G,e} = \frac{M_{G,e}}{V_e - \dfrac{M_{L,e}}{\rho_{L,e}}}$$

From the perfect gas law we can solve for the freon gas pressure

$$P_{G,e} = Z\rho_{G,e} R T_{G,e}$$

The density of the freon liquid varies as a function of the temperature as follows:

$$\rho_{L,e} = 138.84 - 0.105 T_{L,e}$$

Evaporation rate.—Equation (6) is used to solve for the evaporation rate of freon in the flash cooler:

$$W_{G,e} = KVE_e (P_{sat,e} - P_{G,e})$$

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where

$$KVE_e = KrA_e \sqrt{\frac{g_c}{2\pi RT_{L,e}}}$$

The freon saturation pressure $P_{sat,e}$ can be obtained from the saturation curve

$$P_{sat,e} = f(T_{L,e})$$

Heat Exchanger Equations

Energy equation for freon.—The flow throughout the heat exchanger loop is assumed to be equal to the pump flow in the loop. The energy equation for the enthalpy at the heat exchanger exit can then be written as follows:

$$\frac{d(E_x)}{dt} = W_p(H_{L,e} - H_x) + \dot{Q}_x$$

$$H_x = \frac{E_x}{M_x}$$

where M_x is the mass of the fluid inside the heat exchanger and W_p is the pump flow rate. The heat flow from the heat exchanger wall into the freon liquid, $\dot{Q}_x$, is shown in the following equation:

$$\dot{Q}_x = h_x S_x(T_m - T_{L,x})$$

Since freon in the heat exchanger is assumed to be at the saturation pressure of the flash cooler, we have

$$T_{L,x} = T_{L,e} \qquad \rho_{G,x} = \rho_{G,e} \qquad H_{G,x} = H_{V,e}$$

In a two-phase system, X is called the quality of the mixture and is defined as the weight fraction of the material in the vapor phase, or

$$X = \frac{M_G}{M}$$

From the definition of X, we can likewise define the enthalpy of the gas and liquid mixture H_x as

$$H_x = (1 - X)H_{L,x} + XH_{G,x}$$

Rearranging the above equation we can solve for the quality of the fluid in the heat exchanger as follows:

$$X_x = \frac{H_x - H_{L,x}}{\Delta H_x}$$

where

$$\Delta H_x = H_{G,x} - H_{L,x}$$

Let us now solve for the mass of the two-phase fluid inside the heat exchanger. Assume that the mass of the gas in the heat exchanger increases linearly as the fluid flows through the heat exchanger. We can then express the differential mass along the heat exchanger as follows:

$$d(M_G) = \rho_G A_G \, d\ell$$

$$d(M_L) = \rho_L(A_x - A_G)d\ell$$

where

A_G gas flow area
A_x total flow area
$d\ell$ differential length

Since it can be assumed that the gas density is constant along the heat exchanger, we can state that the cross-sectional flow area of the gas for the heat exchanger of length L varies linearly from zero at the entrance, to some value $A_{G,o}$ at the exit. Then at any length ℓ of the heat exchanger,

$$A_G = A_{G,o}\left(\frac{\ell}{L}\right)$$

Now the total mass of the gas in the heat exchanger is

$$M_G = \sum (dM_G) = \int_0^L \rho_G A_G \, d\ell$$

$$= \frac{\rho_G A_{G,o}}{L} \int_0^L \ell \, d\ell$$

$$= \rho_G A_{G,o}\left(\frac{L}{2}\right)$$

Similarly the mass of the liquid in the heat exchanger is

$$M_L = \rho_L L\left(A_x - \frac{A_{G,o}}{2}\right)$$

The total mass of the fluid in the heat exchanger is

$$M_x = M_L + M_G = \rho_L A_x L + A_{G,o}\left(\frac{L}{2}\right)(\rho_G - \rho_L)$$

From the definition of quality, the quality at the heat exchanger outlet is

$$X_o = \frac{dM_{G,o}}{dM_{G,o} + dM_{L,o}} = \frac{\rho_G A_{G,o}}{\rho_G A_{G,o} + \rho_L(A_x - A_{G,o})}$$

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Solving for $A_{G,o}$ we have

$$A_{G,o} = \frac{X_o \rho_L A_x}{\rho_G(1 - X_o) + X_o \rho_L}$$

After substituting for $A_{G,o}$ in the previous equation, the mass of freon in the heat exchanger as a function of quality is

$$M_x = \frac{\rho_L V_x \left[\rho_G \left(1 - \frac{X_o}{2} \right) + \frac{X_o}{2} \rho_L \right]}{\rho_G(1 - X_o) + X_o \rho_L}$$

where

$$V_x = A_x L$$

The following heat transfer coefficient was used for the freon side of the heat exchanger (ref. 1):

$$h_x = 0.0225 \left(\frac{k_x}{D_x} \right) \left(\frac{W_p D_x}{A_x \mu_x} \right)^{0.75} \left[\frac{g_c}{g} \ (J) \ \frac{X_x}{2} \left(\frac{\Delta H_x}{L_x} \right) \right]^{0.375}$$

Heat transfer to the fluid is assumed negligible as it flows from the flash cooler to the heat exchanger. Therefore,

$$H_{L,x} = H_{L,e}$$

$$\Delta H_x = \Delta H_e$$

from the definition of quality we can solve for the gas flow rate:

$$W_{G,x} = X_x W_p$$

Since the pump flow is the total heat exchanger flow, we have for the liquid flow

$$W_{L,x} = W_p - W_{G,x}$$

Energy equation for the heat exchanger wall.—Consider the energy balance for the heat exchanger wall. Heat $\dot{Q}_a$ flows from the tunnel air into the wall and heat $\dot{Q}_x$ flows out of the wall into the freon heat exchanger fluid. The energy equation (2a), has the following form for the wall:

$$\frac{d(T_m)}{dt} = \frac{(\dot{Q}_a - \dot{Q}_x)}{M_m c_{p,m}}$$

the equation for the heat flowing from the tunnel air into the wall can be written as follows:

$$\dot{Q}_a = h_a S_a (T_a - T_m)$$

where

$$T_a = \frac{T_{a,x} + T_{a,f}}{2}$$

the following heat transfer coefficient (ref. 2) was used for the air side of the heat exchanger:

$$h_a = \frac{0.1376 \, k_a}{D_a} \left(\frac{W_a D_a}{A_a \mu_a} \right)^{0.61}$$

Energy equation for the tunnel air.—The tunnel air is divided into two control volumes. One volume $V_{a,x}$ consists of the heat exchanger portion of the tunnel air. The other volume $V_{a,f}$ consists of the remaining tunnel air which includes the tunnel fan where the heat is generated; $T_{a,f}$ is the air temperature at the heat exchanger entrance, and $T_{a,x}$ is the air temperature at the heat exchanger exit. The flow rate in the tunnel circuit, W_a, is assumed to be the same in both control volumes. The energy equation for the heat exchanger control volume can be written as follows:

$$\frac{d(T_{a,x})}{dt} = \frac{W_a(T_{a,f} - T_{a,x}) - \dfrac{\dot{Q}_a}{c_{p,a}}}{M_{a,x}}$$

where

$$M_{a,x} = \rho_a V_{a,x}$$

likewise the energy equation for the fan portion of the control volume can be written as follows:

$$\frac{d(T_{a,f})}{dt} = \frac{W_a(T_{a,x} - T_{a,f}) + \dfrac{\dot{Q}_f}{c_{p,a}}}{M_{a,f}}$$

where

$$M_{a,f} = \rho_a V_{a,f}$$

The heat rate generated by the fan $\dot{Q}_f$ depends on the fan speed.

Pump Flow Loop Equations

Initially this part of the simulation was more detailed but was later simplified because of some occasional computational problems and because it was found that a small variation in pump flow had little effect on the system. The pump flow was therefore made constant, where

$$W_p = 150$$

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Accumulator Equations

Continuity equation and equation of state.—From the continuity equation we have

$$\frac{d(\rho_{ac})}{dt} = \frac{W_{ac} - W_c}{V_{ac}}$$

The pressure in the accumulator can now be obtained from the perfect gas law

$$P_{ac} = \rho_{ac} Z R T_{ac}$$

Orifice equation.—The orifice equation is used to solve for the flow through each control valve unit, or

$$W_V = 0.6 A_V \sqrt{\frac{2 g_c P_{G,e}(P_{G,e} - P_{ac})}{Z R T_{ac}}}$$

where T_{ac}, the temperature in the accumulator, is assumed to be equal to the temperature of the freon gas in the flash cooler. The effective control valve flow area A_V is shown in figure 27. The flow into the accumulator is

$$W_{ac} = N_V W_V$$

where N_V is the number of damper valve units in operation.

Energy equations.—The energy equation for the accumulator is

$$\frac{d(E_{ac})}{dt} = (W_{ac} H_{G,e} - N_c W_{ci} H_{ac})$$

where

W_{ac} flow into the accumulator

$N_c W_{ci}$ total flow leaving the accumulator

$$H_{ac} = \frac{E_{ac}}{\rho_{ac} V_{ac}}$$

$$T_{ac} = f(H_{ac})$$

Compressor Equations

Compressor characteristic curve.—The compressor characteristic curve shown in figure 28 was obtained from a limited amount of available tunnel run data. The ordinate represents the compressor head in thousands of feet and the abscissa is the compressor volumetric flow rate in thousands of cubic feet per minute.

Compressor flow equations.—The equations are

$$\text{HEAD} = Z R T_{ci} \eta_c \left(\frac{\gamma}{\gamma - 1}\right) \left[\left(\frac{P_{co}}{P_{ci}}\right)^{\left(\frac{\gamma - 1}{\eta_c \gamma}\right)} - 1 \right]$$

$$Q = f(\text{HEAD})$$

$$W_{ci} = \frac{Q \rho_{ac}}{60}$$

The compressor outlet flow is equal to the compressor inlet flow plus the vapor flow from the economizer, or

$$W_{co} = W_{ci} + W_{G,ec}$$

Line pressure drops from the accumulator to the compressor inlet and from the compressor outlet to the condenser are neglected, or

$$P_{ci} = P_{ac}$$

$$P_{co} = P_{con}$$

There is also no heat transfer from the accumulator up to the compressor inlet, or

$$T_{ci} = T_{ac}$$

Figure 27.—Control valve area.

Figure 28.—Compressor characteristic curve; speed, 6650 rpm.

Compressor energy equations.—The compressor is divided into two segments. All of the economizer gas is assumed to discharge into the compressor at the entrance of segment 2. The equation below is the temperature rise of the fluid for compressor segment 1:
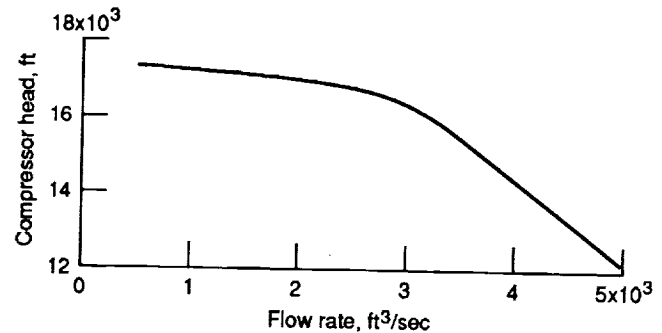
$$T_{c1} = T_{ci} \left(\frac{P_{ec,d}}{P_{ci}} \right)^{\frac{\gamma-1}{\eta_c \gamma}}$$

Since $dH_{\text{actual}} = 1/\eta_a (dH_{\text{isentropic}})$ and since $dH = c_p \, dT$, we have for the enthalpy

$$H_{c1} = H_{ac} + \left(\frac{c_{p,G}}{\eta_a} \right) (T_{c1} - T_{ci})$$

The vapor from the economizer with enthalpy $H_{V,ec}$ is returned to the compressor and mixes with the compressor fluid of segment 1. From the energy equation, we can express the enthalpy of the mixture as follows:

$$H_{c2} = \frac{W_{ci} H_{c1} + W_{G,ec} H_{V,ec}}{W_{co}}$$

or the temperature of the mixture

$$T_{c2} = \frac{W_{ci} T_{c1} + W_{G,ec} T_{L,ec}}{W_{co}}$$

Fluid with enthalpy H_{c2} and temperature T_{c2} enters the second segment of the compressor and rises to a temperature T_{co} at the compressor exit, or

$$T_{co} = T_{c2} \left(\frac{P_{co}}{P_{ec,d}} \right)^{\frac{\gamma-1}{\eta_c \gamma}}$$

The enthalpy at the compressor exit then is

$$H_{co} = H_{c2} + \left(\frac{c_{p,G}}{\eta_a} \right) (T_{co} - T_{c2})$$

Condenser Equations

Energy equations.—The energy equation for the condenser can be written as follows:

$$\frac{d(E_{L,\text{con}})}{dt} = W_{\text{con}} (H_{G,\text{con}} - H_{L,\text{con}}) - \dot{Q}_{\text{con}}$$

$$H_{L,\text{con}} = \frac{E_{L,\text{con}}}{M_{L,\text{con}}}$$

where the flow into the condenser is assumed to be the flow out of the compressor W_{co}. Also the enthalpy H_{co} out of the compressor is assumed to be the same as the enthalpy flowing into the condenser. The variables W_{con} and $H_{L,\text{con}}$ are, respectively, the flow and the enthalpy of the liquid freon leaving the condenser.

By neglecting the heat capacity of the condenser piping, the heat rate removed from the condenser by the water, $\dot{Q}_{\text{con}}$, can be written as follows:

$$\dot{Q}_{\text{con}} = h_{\text{con}} S_{\text{con}} (T_{L,\text{con}} - T_w)$$

The average condenser water temperature T_w is calculated as follows:

$$T_w = 0.5 (T_{wi} + T_{wo})$$

The heat transfer coefficient for the condenser is too complex to be calculated analytically. An overall heat transfer coefficient $h_{\text{con},0}$ was calculated based on results of known system operating conditions. The 0 assigned to variables represents tunnel run data which was obtained either directly or calculated from data. This data was used to calculate $h_{\text{con},0}$. The equations used in the calculation $h_{\text{con},0}$ are based on the steady-state part of the energy equation (2) and also equation (3), or

$$\dot{Q}_{\text{con},0} = h_{\text{con},0} S_{\text{con}} (T_{L,\text{con},0} - T_{w,0})$$

But in steady state the heat rejected at the condenser is equal to the heat added by the compressor plus the heat added in the tunnel, or

$$\dot{Q}_{\text{con},0} = \dot{Q}_{c,0} + \dot{Q}_{f,0} = W_{\text{con},0} (H_{co,0} - H_{G,e,0}) + \dot{Q}_{f,0}$$

solving for $h_{\text{con},0}$ we have

$$h_{\text{con},0} = \frac{\dot{Q}_{f,0} + (H_{Co,0} - H_{G,e,0}) W_{\text{con},0}}{S_{\text{con}} (T_{L,\text{con},0} - T_{w,0})}$$

A heat transfer coefficient was used which is a function of the cube root of condenser flow, or

$$h_{\text{con},0} = K_h W_{\text{con},0}^{0.333}$$

Using the above equation to solve for K_h gives

$$K_h = \frac{h_{\text{con},0}}{W_{\text{con},0}^{0.333}}$$

The value of K_h calculated above from tunnel run data is now used to calculate h_{con} for all other operating conditions, or

$h_{con} = K_h W_{con}^{0.333}$. Or by eliminating K_h we can write the heat transfer coefficient in the condenser as follows:

$$h_{con} = h_{con,0} \left(\frac{W_{con}}{W_{con,0}} \right)^{0.333}$$

The freon leaving the condenser is assumed to be saturated liquid. We can therefore obtain the corresponding temperature of the liquid from the saturation curve for freon, or

$$T_{L,con} = f(H_{L,con})$$

The energy equation for the water leaving the condenser is

$$\frac{d(T_{wo})}{dt} = \frac{W_w (T_{wi} - T_{wo}) + \dfrac{\dot{Q}_{con}}{c_{p,w}}}{V_w \rho_w}$$

where W_w is the water flow rate through the condenser and T_{wo} is the outlet water temperature for the condenser. The inlet water temperature T_{wi} is assumed to be constant.

Continuity equation.—From the continuity equation

$$\frac{d(M_{con})}{dt} = W_{co} - W_{con}$$

where M_{con} is the total mass of freon inside the condenser and W_{con} is the liquid freon leaving the condenser (assumed equal to the freon condensation rate).

The relative volume occupied by the gas and liquid inside the condenser is assumed to remain constant (i.e., $V_{L,con}$ is constant). The mass of the gas and the liquid freon inside the condenser can then be obtained from the following two equations:

$$M_{L,con} = V_{L,con} \rho_L$$

$$M_{G,con} = M_{con} - M_{L,con}$$

Freon condensation rate.—Equation (6) is used to solve for the condensation rate of freon in the condenser, or

$$W_{con} = KVE_{con}(P_{con} - P_{sat,con})$$

where

$$KVE_{con} = KrA_{con}\sqrt{\frac{g_c}{2\pi RT_{L,con}}}$$

Equation of state.—From the definition of density we can obtain the density of the gas as follows:

$$\rho_{G,con} = \frac{M_{G,con}}{V_{G,con}}$$

From the perfect gas law, we can now solve for the pressure of the freon inside the condenser, or

$$P_{con} = \rho_{G,con} ZRT_{G,con}$$

$P_{sat,con}$ is the saturation pressure of the freon in the condenser. It can be obtained from the freon saturation curve of the PH diagram:

$$P_{sat,con} = f(T_{L,con})$$

Economizer equations

The system uses two economizer stages where part of the freon liquid is flashed to gas and returned to the inlet of compressor stages 2 and 3. For simplicity, these will be approximated in the simulation by one economizer only. Because the fluid volumes in the economizers are relatively small, the transients in the economizer are neglected.

Flow equation.—The economizer gas is returned to the center stage of the compressor. The average pressure at the center stage is estimated as follows, where K_{ed} is a constant:

$$P_{ec,d} = K_{ed}P_{co} + (1 - K_{ed})P_{ac}$$

The frictional loss of the economizer gas discharge flow is assumed to be negligible. Consequently, the pressure P_{ec} inside the economizer is assumed to be equal to the discharge pressure at the compressor $P_{ec,d}$, or

$$P_{ec} = P_{ec,d}$$

Since the fluid in the economizer is at saturation, it is assumed that the economizer saturation pressure $P_{sat,ec}$ is equal to P_{ec}:

$$P_{sat,ec} = P_{ec}$$

Since economizer transients are neglected, the steady-state continuity equation results in

$$W_{L,ec} = W_{con} - W_{G,ec}$$

Energy equation.—From the steady-state energy equation, we can write for the economizer

$$W_{con}H_{L,con} - W_{G,ec}H_{G,ec} - W_{L,ec}H_{L,ec} = 0$$

Substituting the previous expression for $W_{L,ec}$ and after rearranging, the freon vapor returning to the compressor is

$$W_{G,ec} = \frac{W_{con}(H_{L,con} - H_{L,ec})}{H_{G,ec} - H_{L,ec}}$$

The fluid inside the economizer is assumed to be in the saturated region. The freon temperature and the freon enthalpies of the liquid and gas can then be obtained from the saturation curves, or

$$T_{L,ec} = f(P_{sat,ec})$$

$$H_{L,ec} = f(T_{L,ec})$$

$$H_{G,ec} = f(T_{L,ec})$$

Derivation of Evaporation Equation (from ref. 3)

A nonrigorous derivation of the equation used in calculating the evaporation flow is given here. Because of the complexity of the molecular motion, this equation cannot give good quantitative results. The equation is used mainly to account for the transient dynamics of evaporation in the flash cooler and the condenser. According to gas kinetic theory the number of molecules striking a unit area per unit time is

$$N_T = \frac{n}{V}\left(\frac{C_a}{4}\right)$$

where

n/V number of molecules per volume
C_a mean velocity of the molecules

The mass of the molecules striking a surface area A per unit time can be expressed as follows:

$$W = AN_T\left(\frac{mw}{Na}\right) = A\left(\frac{n}{V}\right)\left(\frac{C_a}{4}\right)\left(\frac{mw}{Na}\right)$$

where

Na Avogadro's number
m mass of molecule $= mw/Na$
mw molecular weight

By expressing the gas density ρ as follows,

$$\rho = \frac{M}{V} = \frac{nm}{V}$$

we can now rewrite the mass flow rate W as

$$W = A\rho\left(\frac{C_a}{4}\right)$$

From gas kinetic theory, the kinetic energy of a gas molecule is

$$KE = \left(\frac{3}{2}\right)k_B T$$

where T is the temperature and k_B is the Boltzmann constant. From the perfect gas law $PV = nk_B T$; kinetic energy KE is then equal to

$$KE = \left(\frac{3}{2}\right)P\left(\frac{V}{n}\right)$$

and from the definition of kinetic energy we have

$$KE = m\left(\frac{C^2}{2g_c}\right)$$

Then equating the two kinetic energy equations gives $(3/2)P(V/n) = mC^2/(2g_c)$, and substituting for $n(m/V) = M/V = \rho = P/(RT)$ and rearranging we have

$$C = \sqrt{3g_c RT}$$

The mean velocity C_a is related to the root mean square velocity C as follows:

$$C_a = \sqrt{\left(\frac{8/3}{\pi}\right)}C$$

or

$$C_a = \sqrt{8g_c R\frac{T}{\pi}}$$

Now since $W = A\rho(C_a/4) = A(P/R/T)(C_a/4)$ and after substituting for C_a and rearranging we have for the mass flow rate

$$W = PA\sqrt{\frac{g_c}{2\pi RT}}$$

The preceding equation describes the ideal gas condensation rate. It assumes that all of the molecules that strike the surface of the liquid will condense. In actuality some of the molecules will be reflected back. To account for this, the equation is modified as follows:

$$W = KrAP\sqrt{\frac{g_c}{2\pi RT}}$$

where the reflection coefficient Kr is the fraction of the molecules that hit the surface and are absorbed by the liquid. Now suppose we have a closed vessel where the liquid resides. Then at equilibrium the condensation rate W_{con} is equal to the evaporation rate W_{ev}. Let us suppose further that the evaporation flow rate can be expressed similarly except that T is the temperature of the liquid and P is the saturation pressure of the liquid. We can now express the net flow between the gas and the liquid as follows:

$$W = W_{ev} - W_{con}$$

$$= KrAP_{sat}\sqrt{\frac{g_c}{2\pi RT_L}} - KrAP_G\sqrt{\frac{g_c}{2\pi RT_G}}$$

$$= KrA\sqrt{\frac{g_c}{2/\pi/R}}\left(\frac{P_{sat}}{\sqrt{T_L}} - \frac{P_G}{\sqrt{T_G}}\right)$$

Since the temperature of the gas T_G can be assumed to be equal to the temperature of the liquid T_L, we then have

$$W = KrA\sqrt{\frac{g_c}{2\pi RT_L}}(P_{sat} - P_G)$$

Appendix B
Symbols

A	flow area, ft^2		P_G	freon gas pressure, psf
A_v	total valve area, ft^2		P_{sat}	saturation pressure, psf
A_x	freon flow area in heat exchanger		P	pressure, psf
C	root mean square velocity, ft/sec		Q	volumetric flow rate, ft^3/min
C_a	mean velocity, ft/sec		$\dot{Q}$	heat flow rate, Btu/sec
c_p	specific heat, Btu/lbm-°R		R	freon gas constant, 12.776 ft-lbf/lbm-°R
D	diameter, ft		S	surface area, ft^2
$d\ell$	differential length		T	temperature, °R
e	error		t	time, sec
er	error (limiter output)		V	volume, ft^3
FR_{con}	fraction of condenser volume occupied by gas		$V_{a,f}$	tunnel air volume outside of heat exchanger, ft^3
f(HEAD)	compressor characteristic curve		$V_{a,t}$	total tunnel air volume, ft^3
G	controller gain		$V_{a,x}$	tunnel air volume assigned to heat exchanger, ft^3
g	gravitational constant, ft/sec^2		V_x	freon volume in heat exchanger, ft^3
g_c	conversion constant, 32.2 lbm-ft/lbf-sec^2		W	flow rate, lbm/sec
H	enthalpy, Btu/lbm		X	quality, dimensionless
H_V	enthalpy of saturated vapor, Btu/lbm		Z	gas compressibility factor, dimensionless
HEAD	ft-lbf/lbm		ΔH	heat of vaporization, Btu/lbm
h	heat transfer coefficient, Btu/sec-ft^2-°R		γ	specific heat ratio, dimensionless
J	conversion constant, 778 ft-lbf/Btu		η_a	adiabatic efficiency
$K_{e,d}$	flash cooler flow discharge constant		η_c	compressor efficiency
K_f	friction constant		μ	viscosity, lbm/ft-sec
KE	kinetic energy		ρ	density, lbm/ft^3
Kr	fraction of molecules absorbed by liquid			
KS	controller constant		Subscripts:	
KT	controller gain constant		a	air; adiabatic
KV	controller gain constant		ac	accumulator
k	thermal conductivity, Btu/sec-ft-°R		c	compressor
k_B	Boltzmann constant		ci	compressor inlet
L	length, ft		co	compressor outlet
ℓ	variable length of heat exchanger, ft		con	condenser
M	mass, lbm		$c1$	compressor segment 1
M_k	controller output		$c2$	compressor segment 2
m	mass of molecule, mw/Na		d	discharge
mw	molecular weight		e	flash cooler (evaporator)
N_c	number of compressor units in operation		ec	economizer
N_p	pump speed, rpm		ev	evaporator
N_v	damper valve units		f	fan
Na	Avogadro number		G	gas
n	number of molecules		i	inlet
$P_{ec,d}$	economizer discharge pressure, psf		$k-n$	$(k-n)^{th}$ sampling constant

L	liquid	w	water
m	heat exchanger metal	wi	water in
o	outlet	wo	water out
p	pump	x	heat exchanger
sat	saturation	0	tunnel run data or calculated from tunnel run data
V	valve		
v	vapor		

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16. Abstract

The Icing Research Tunnel (IRT) at the NASA Lewis Research Center is a subsonic, closed-return atmospheric tunnel. The tunnel includes a heat exchanger and a refrigeration plant to achieve the desired air temperature and a spray system to generate the type of icing conditions that would be encountered by aircraft. At the present time, the tunnel air temperature is controlled by manual adjustment of freon refrigerant flow control valves. An upgrade of this facility calls for these control valves to be adjusted by an automatic controller. This report discusses the digital computer simulation of the IRT refrigeration plant and the automatic controller that was used in the simulation.

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